

PRE-STRESS MODAL ANALYSIS OF A CENTRIFUGAL PUMP IMPELLER FOR DIFFERENT BLADE THICKNESSES

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ABSTRACT

Centrifugal pump impellers are high speed rotating components vulnerable to vibration resulting in the failure of the system pump eventually. While rotating, stresses are created due to the inertial forces. The model characteristics change due to the stress created and hence it is important to measure the model characteristics with consideration to the prestresses. Therefore, in this paper, the prestressed modal analysis of the impeller with blade thicknesses ranging from 3mm to 5mm with increments of 0.5mm, are analyzed by the numerical method, FEA for a rotational velocity of 3600rpm. The experimental modal testing of the impellers is then done using electro-dynamic shaker, to validate the FEA results.

KEYWORDS: Pre-Stress Modal Analysis, Impeller, Mode Shape, Natural Frequency, Von-Mises Stress & Blade Thickness

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INTRODUCTION

Modal analysis plays an important role in the design and optimization of dynamic structures. Modal analysis is the method of determining the natural frequency and mode shapes of a component that, it may have when it is subjected to resonance conditions. Modal characters are inherent of the component. Centrifugal pump impellers are high speed rotating components, that have a high chance of rotor failure by attaining resonance frequencies. While rotating, stresses are created in the impeller due to its inertial effects which may change the inherent character of the impeller namely the natural frequencies and mode shapes. Therefore, the modal testing may be done considering the prestresses for accurate results. A parametric study of a centrifugal pump impeller, by changing the outlet blade angle using the numerical method of computational fluid dynamics [1,2], discretized three-dimensional, incompressible Navier-Stokes equations over an unstructured grid was accomplished, with a commercial computational fluid dynamics finite-volume code. For each impeller, the flow pattern and the pressure distribution in the blade passages were calculated, and finally, the head-capacity curves were compared and discussed [3,4].

Transverse vibration characteristics of a thin spinning disc under free and forced vibrating conditions, using the numerical method found that the natural frequencies of the thin spinning plate attached to a rigid core increased significantly, with an increase in the speed [5,6]. The modal properties of the fluid-structure system by an experimental modal analysis with the impeller suspended in air and inside a water reservoir can be analysed [7,8]. The impeller was excited with an instrumented hammer, and the response measured by means of miniature

accelerometers. The Frequency Response Functions (FRFs) have been obtained from a large number of impacting positions, in order to ensure the identification of the main mode shapes. As a result, the main modes of vibration have been well characterized both in air and in water, in terms of natural frequency, damping ratio and mode shape.

Numerical simulation carried out to analyse modal behavior of a reduced scale pump-turbine impeller. The simulation has been done using the numerical method, in air and in water [9]. The same boundary conditions in the experiment were considered: free body in air and free body submerged in a reservoir of water. The stress distribution and concentration of a single blade in the impeller, under the different pressure fields of the blade structure surface, and also studied the fatigue failure of the blades [10]. A method developed for the performance computation of an expandable-impeller pump and validated it [11,12]. Numerical method was used to estimate the total radial load on the impeller of two test pumps as a function of flow rate. The predictions were validated with the experimental data of global characteristics, and unsteady pressure distribution round the impeller. Structural stress analysis and modal analysis carried out by the researchers, on an impeller by One-way Fluid Structure Interaction technique in the numerical method [13,14]. The study showed that the maximum equivalent stress occurs at the liner near the blade pressure surface of the impeller. The experimental modal analysis of a fan by using the hammer-hitting pulse-inspirit method revealed that the natural frequencies of the fan vibration.

Dynamics analysis of the pump and turbine of hydrodynamic coupling, the modal vibration analysis data of pump and turbine could be obtained by the numerical method, and its natural frequencies at different rotational speeds are calculated [15]. All the above survey has not considered the exact impeller geometry shape, i.e. all the earlier researchers assumed uniform blade thickness, like disk. This work addresses the exact impeller geometry shape with different thicknesses, and is concerned with the numerical method and modal testing of an industrial impeller. The objective is to determine and verify the vibration characteristics of the impeller, using both the numerical and experimental methods.

STATIC STRUCTURAL ANALYSIS

The 3D model for the finite element analysis is generated, using the Bladegen feature in the ANSYS-Workbench. The impeller is a shrouded one, designed for a radial type centrifugal pump. The assumptions and the impeller parameters are given in Table 1. Figure 1 shows that the three-dimensional model of the impeller, using the Bladegen feature in the ANSYS Workbench. The finite element model of the impeller is prepared and then analysed, using the ANSYS Workbench. The impeller is composed of a complex geometric structure, and hence, for better results tetrahedral elements are used. Impellers are high speed rotating components that may have considerable inertial effects on the modal behavior of the impeller. Therefore, the modal analysis may be done considering the stress effects due to rotation. A rotating velocity of 3600 rpm is given to the impeller in the clockwise direction. The static structural analysis is conducted by applying the rotational velocity on the impeller. The stresses acting on the impeller due to the inertial forces created due to rotation are determined. The stress values obtained from the stress analysis are used as pre-stress values for the modal analysis.

Table 1: Impeller Parameters

Type	Radial, Closed impeller
Fluid	Water
Flow rate	60 m ³ /hr
Head	51m

Table 1: Contd.,	
Number of blades	6
Blade inlet angle	35 degrees
Blade exit angle	26 degrees
Suction diameter	85mm
Impeller diameter	215mm
Material	Cast Iron
Rotating velocity	3600rpm

MODAL ANALYSIS USING FEM

The modal analysis is done on five impellers with different blade thicknesses, ranging from 3.0 mm to 5.0 mm, with increments of 0.5mm. Six modes were extracted for each impeller. The natural frequencies of the extracted modes range from 450 to 2200 Hz. Figures 3 to 8 show the mode shapes of the impeller obtained from the modal analysis for an impeller, with the blade thickness of 4.0 mm. In modes 1 and 2, the nodal diameter is found to be one. Zero nodal diameters are exhibited by the impeller in modes 3 and 4. Modes 5 and 6 exhibited nodal diameters of two. Though the natural frequencies of the impeller are found to change considerably with the blade thickness of the impeller, the mode shapes did not show a notable change with the change in the blade thicknesses



Figure 1: Three-Dimensional Geometric Model for Thickness $T = 3.0$ Mm

The points, which remain stationary in the vibration cycle, form a diametric line called nodal diameter. Natural frequency is the frequency at which an object vibrates when excited by force. At this frequency, the structure offers the least resistance to a force, and if left uncontrolled, failure can occur. The mode shape is a deflection of an object at a given natural frequency. The eigenvalue (natural frequency) and the accompanying eigenvector (mode shape) are calculated to define the dynamics of a structure. The term nodal diameter is derived from the appearance of a circular geometry, like a disk, vibrating in certain mode.

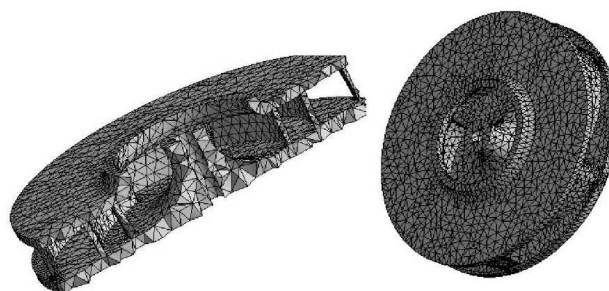


Figure 2: Finite Element Model of the Impeller with Tetrahedral Elements

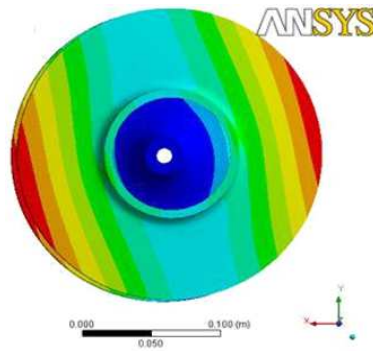


Figure 3: Mode Shape 1 for the Impeller with Blade Thickness $T=4.0$ Mm

Figures 3 to 8 show the various mode shapes for the extracted modes. It is noted that even though the natural frequencies of the impeller change considerably as the thickness changes, there was not much change in the mode shapes of the impellers. These mode shapes obtained for the impeller analysed, show congruence with the results obtained by Ziaei Rad [16].

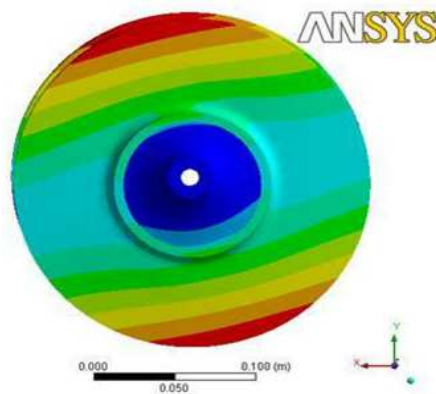


Figure 4: Mode Shape 2 for the Impeller with Blade Thickness $T=4.0$ Mm

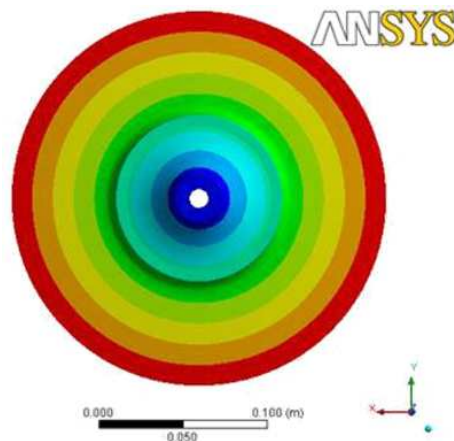


Figure 5: Mode Shape 3 for the Impeller with Blade Thickness $T=4.0$ Mm

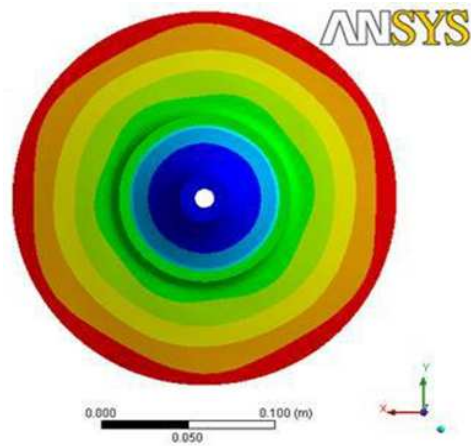


Figure 6: Mode Shape 4 for the Impeller with Blade Thickness $T=4.0$ Mm

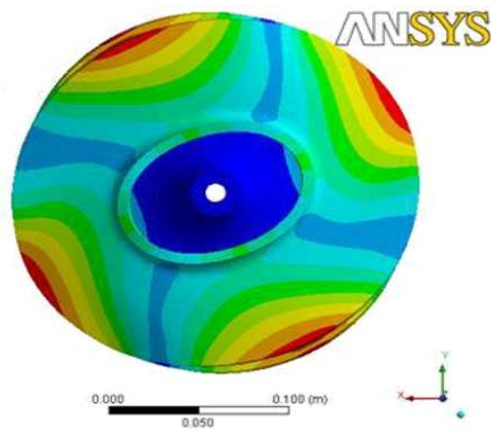


Figure 7: Mode Shape 5 for the Impeller with Blade Thickness $T=4.0$ Mm

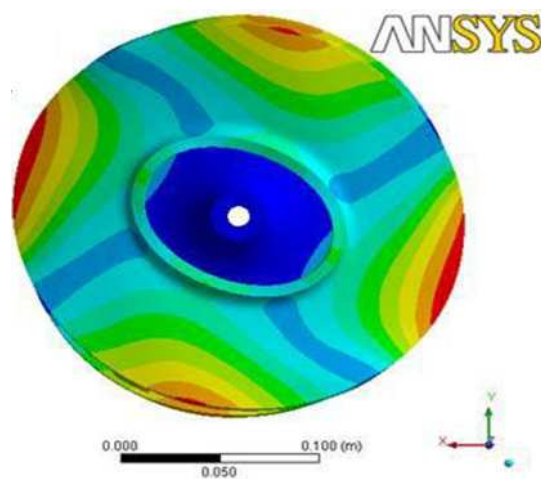


Figure 8: Mode Shape 6 for the Impeller with Blade Thickness $t=4.0$ mm

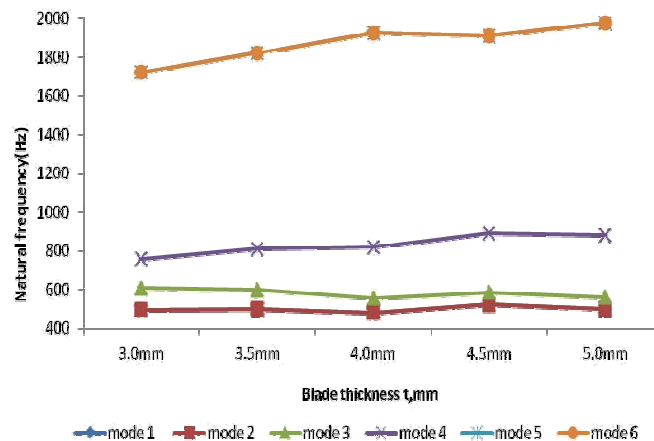


Figure 9: Natural Frequency from Modal Analysis Vs Blade Thickness

It is also noted that the natural frequencies exhibited by the impeller in modes 1 and 2 are almost the same as those with similar mode shapes, with nodal diameter one and no node circles formed. While the natural frequency increased with respect to the blade thickness, the natural frequency showed a gradual drop with an increase in the blade thickness for mode 3. Modes 3 and 4 have a nodal circle. Modes 5 and 6 also have the same frequency values, without any notable differences in the frequency values, as well as the mode shapes with nodal diameter two. It is found that, as the thickness of the impeller increases the natural frequency also increases. Figure 9 shows the plot between the natural frequency and the blade thicknesses of the impeller for different modes. The values of mode 1 and mode 2 are very close to each other, and the values of mode 5 and 6 are close to each other, as shown in the above-mentioned figures.

EXPERIMENTAL PRESTRESS MODAL ANALYSIS TESTING

The natural frequencies obtained from the finite element analysis are validated by the experimental modal testing. For the testing, the impeller was fabricated by casting. The modal testing was done for impellers of different thicknesses. The impeller is constrained at the center. An electric motor is used to excite the structure. The transducer attached to the shaker measures the input load that is given. The output response is measured using an accelerometer attached to the impeller. The input impulse and output response signals are fed into a signal conditioner. The signals are then processed by an FFT analyzer, which in turn, is connected to a computer system, which acquires and processes the digital frequency domain data into frequency response functions. The natural frequencies are then obtained from the FRF data.

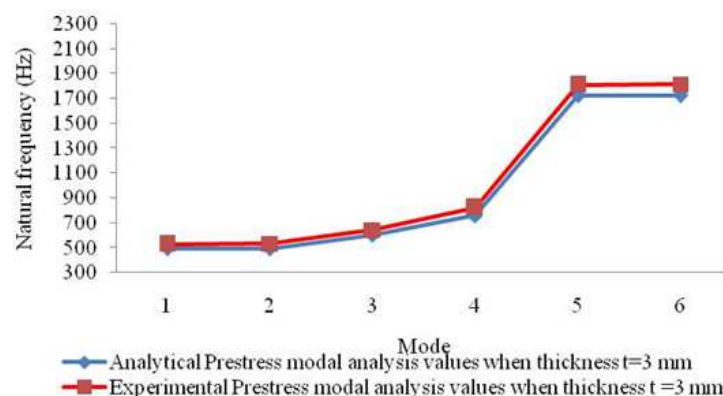


Figure 10: Mode Vs Natural Frequency Comparison of the Experimental Prestress Modal

Value and FEA Values for Blade Thickness T=3.0 Mm

VALIDATION OF THE FEA PRESTRESS MODAL ANALYSIS VALUES WITH THE EXPERIMENTAL PRESTRESS MODAL ANALYSIS VALUES

The values obtained in the testing are very close to the FEA results. Figures 10 to 14 are the natural frequency values obtained from the experimental modal testing and the FEA plotted against the modes. It can be noticed that the natural frequency values obtained from the experiment and the FEA are very close, thus validating the results obtained.

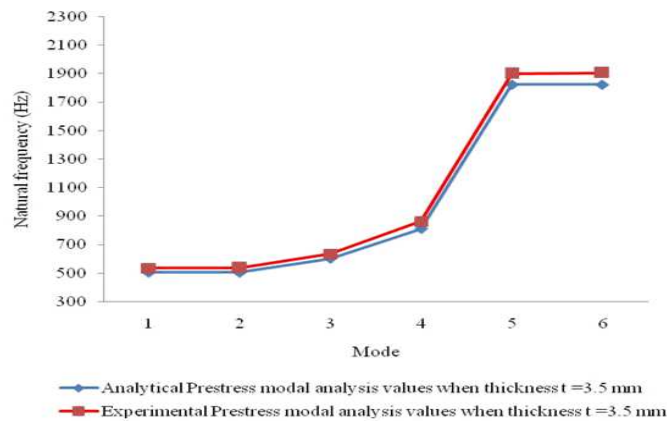


Figure 11: Mode Vs Natural Frequency Comparison of the Experimental Prestress

Modal Value and FEA Values for Blade Thickness T=3.5 Mm

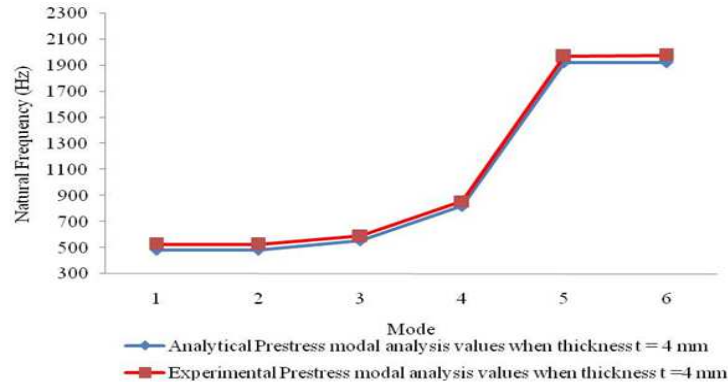


Figure 12: Mode Vs Natural Frequency Comparison of the Experimental Prestress

Modal Value and FEA Values for Blade Thickness T=4.0 Mm

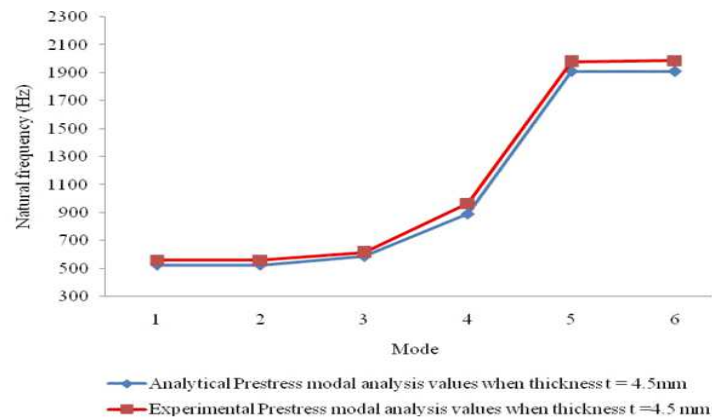


Figure 13: Mode Vs Natural Frequency Comparison of the Experimental Prestress

Modal Value and FEA Values for Blade Thickness T=4.5 Mm

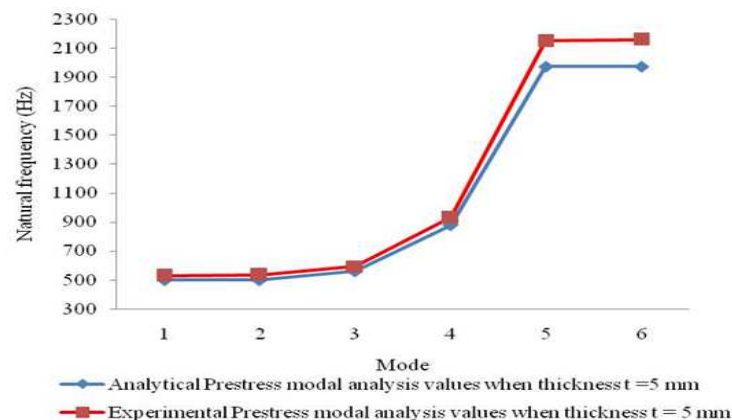


Figure 14: Mode Vs Natural Frequency Comparison of the Experimental Prestress

Modal Value and FEA Values for Blade Thickness T=5.0 Mm

The prestress modal analysis of the impellers ranging from 3.0 mm to 5.0 mm is done by the Finite Element Analysis at a rotational velocity of 3600rpm. The natural frequencies and the mode shapes of the impellers are extracted. It is found that the natural frequency of the impeller increased considerably, as the thickness of the impeller blades increased. This increase in the natural frequency can be attributed to the increase in the stiffness of the structure overcoming the effect of the addition of mass. Even though the thickness has a considerable effect on the natural frequencies, the mode shapes of the impeller at resonance are not affected due to the increase in thickness. The results obtained from the FEA were validated by conducting the experimental modal testing on the impellers, by the shaker testing method.

CONCLUSIONS

The prestressed modal analysis of the impellers ranging from 3.0mm to 5.0mm was done by the Finite Element Analysis at a rotational velocity of 3600rpm. The natural frequencies and the mode shapes of the impellers were extracted. It was found that the natural frequency of the impeller increased considerable as the thickness of the impellers blades increases. This increase in natural frequency can be attributed to the increase in the stiffness of the structure overcoming the effect of the addition of mass. Even though the thickness had a considerable effect on the natural frequencies, the mode

shapes of the impeller at resonance were not affected due to the increase in thickness. The results obtained from the FFA were validated by conducting the experimental model testing on the impellers, by the shaker testing method.

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